

Estimation Algorithms for Low Pressure Cooled EGR in Spark-Ignition Engines

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ABSTRACT

Low-pressure, Cooled Exhaust Gas Recirculation (LPC EGR) brings significant fuel economy, NO_x reduction and knock suppression benefits to a modern, boosted, downsized Spark Ignition (SI) engine. As a prerequisite to design an engine control system for LPC EGR, this paper presents development of a set of estimation algorithms to accurately estimate the flow rate, pressure states and thermal states of the LPC EGR-related components.

CITATION: Liu, F. and Pfeiffer, J., "Estimation Algorithms for Low Pressure Cooled EGR in Spark-Ignition Engines," *SAE Int. J. Engines* 8(4):2015, doi:10.4271/2015-01-1620.

INTRODUCTION

For diesel engines, Low-pressure Cooled Exhaust Gas Recirculation (LPC EGR) has long been recognized as an effective way to mitigate engine-out NO_x and hence has gained significant popularity in production applications over the years. During the last decade, the benefit of LPC EGR on spark-ignited (SI) combustion has also attracted a collective effort in industry and academia to explore its implementation on SI engines, especially its ability to reduce exhaust gas temperature (to reduce fuel enrichment needed to protect the exhaust components) and suppress knocking. A substantial number of studies have concluded that LPC EGR is an effective way to meet the demands of highly boosted and downsized engine designs that aim to meet current and future stringent fuel economy and emission standards [1, 2, 3, 4].

While the benefits of LPC EGR drive engine and vehicle manufacturers' desire to integrate LPC EGR into production engine applications, the control of LPC EGR poses a significant challenge to the Engine Management System (EMS) design. One significant reason for the challenge is the lack of a low cost, accurate EGR rate sensor. For lab tests and research activities, the EGR rate can be determined by measuring the intake manifold CO₂ concentration (usually done by an emissions measurement instrument). Alternately, a wide-range air fuel ratio sensor could be used to measure the oxygen content in the intake gas flow as a method to calculate the EGR rate. However, wide range air fuel ratio sensors can be susceptible to water impact, leading to thermal cracking.

Without having an accurate measurement of actual EGR rate, the EGR control algorithm must be conservative with regard to scheduling desired EGR rate. The control may not realize the full benefit of EGR (e.g. only scheduling up to 10%, due to the achievable estimation accuracy of the EGR rate estimation algorithm [5]).

However, many researchers have shown that a more aggressive EGR rate (i.e. up to 20-25% [1, 2, 3, 4, 5]) can achieve greater benefit and still be within combustion stability limitations.

If a heavy EGR rate is scheduled but the EGR rate is not accurately estimated, this can lead to poor control. EMS control logic depends on the EGR rate to determine the optimal engine operating conditions during heavy EGR operation. Some of the important control parameters during heavy EGR operation include the fuel injection mass, spark advance timing, throttle opening, valve train phasing, etc. Without proper control, good combustion cannot be maintained, leading to loss of torque, knocking or even misfire.

Hence, it is very important to develop a LPC EGR flow rate estimation method that can estimate EGR rate with sufficient accuracy. The accurate EGR rate estimate allows an aggressive EGR target to be scheduled confidently by the EMS control algorithm to fully realize the benefits from LPC EGR.

In this paper, development work is presented for an estimation method of LPC EGR flow rate, as part of a production-intent LPC EGR EMS system for a boosted SI engine. This includes related and supporting quantities for LPC EGR flow estimation, including EGR/inlet air mixing point pressure and temperature estimation, and EGR valve upstream pressure. The achieved EGR estimation accuracy is demonstrated by the engine dynamometer test results. The EGR estimation accuracy, particularly the sensitivity to the differential pressure across the EGR valve, is also analyzed.

ENGINE CONFIGURATION AND TEST SETUP

Figure 1 shows the mechanization of the investigated engine

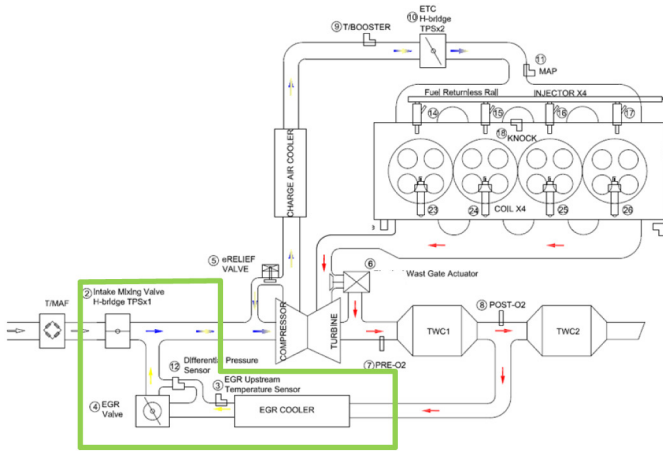


Figure 1. Engine setup with LPC EGR hardware (highlighted).

The tested engine is a multiple port fuel injection (MPFI) turbocharged 4-cylinder SI engine. Besides the typical components found on a turbocharged PFI engine, the LPC EGR hardware consists of an EGR cooler, an EGR valve, a differential pressure sensor, EGR gas temperature sensor, mass air flow meter (MAF) and an inlet mixing valve.

The LPC EGR valve is a butterfly plate type, DC motor actuated, single position feedback valve. The EGR cooler has substantial heat rejection capability, cooling the EGR gas temperature from more than 400 degrees Celsius down to near engine coolant temperature. The EGR gas temperature sensor measures the temperature after the EGR cooler and into the EGR valve. The pressure drop across the EGR valve is measured by the differential pressure sensor.

There have been multiple studies [6,7] comparing the advantages and disadvantages of different possible EGR routing options. In this test engine setup, the EGR gas is sourced after turbine and closed-coupled catalytic converter. Such a setup benefits from favorable thermal properties and “cleaner” gases. With this configuration, the EGR is introduced before the compressor. Therefore, the pressure difference between the exhaust pipe and the intake path is positive and creates a natural driving force for the EGR gases.

However, this setup also causes the case where the engine exhaust back pressure can be insufficient to drive the desired amount of EGR through the LPC EGR loop. Therefore, especially under low speed or/and low load condition, it is necessary for the inlet mixing valve (IMV) to be closed. By closing the IMV, a small amount of vacuum is created at the EGR/air mixing point. This vacuum will then create the necessary amount of positive pressure drop across the EGR valve, to draw the desired amount of EGR flow into the induction system. Tests have shown that a very minimal amount of vacuum, in a range of 1 kPa, will be sufficient to drive more than 20% EGR into the induction system. The decision to close the inlet mixing valve also depends on the available exhaust back pressure, which could be

different from one LPC EGR routing option to another (i.e. choice of sourcing EGR before or after turbine, etc.). Some diesel applications also use a back pressure valve instead of an inlet mixing valve. However, the much higher exhaust gas temperature on a SI engine can pose a challenge for that option.

It is also worth noting that the ignition coil has been updated in this test engine to 105 mJ Delphi high ignition energy coil family, up from the base coil of 70 mJ. This increased level of ignition energy is necessary to sustain a good combustion quality during heavy EGR operation.

To examine the EGR rate estimation accuracy, the reference EGR rate is measured by two means in this study:

- CO₂ concentration measured at intake manifold plenum location and exhaust manifold location by lab grade emission bench
- O₂ concentration measured at pre-ETC location (immediately before throttle) and exhaust manifold location, by Delphi OSL family production Wide-Range Air Fuel ratio (WRAF) sensors.

The reference EGR rate is obtained by the following calculation methods, with these guidelines:

- these equations are for mole-based EGR percentage;
- the calculation of mass-based EGR percentage will be slightly different;
- the difference is considered negligible for the context of the production EMS applications

$$EGR \% = \frac{CO_2 \%_{intake}}{CO_2 \%_{exhaust}} \quad (1)$$

$$EGR \% = \frac{O_2 \%_{air} - O_2 \%_{intake}}{O_2 \%_{air} - O_2 \%_{exhaust}} \quad (2)$$

The nominal 20.8% is used as the standard O₂ percentage in air. For stoichiometric combustion in an SI engine, the amount of O₂ present in the engine-out exhaust gas is only a trace amount, thus it is usually acceptable to assume zero in the context of the EGR fraction calculation. Figure 2 shows an example of EGR measurement based on the above two methods, while the EGR valve was cycling between two positions.

In this study, the test shows the WRAF sensor-based EGR measurement correlates well with the emission bench-based EGR measurement, with a nominal difference within 1 % EGR rate, as shown in Figure 2. Note the emission bench-based EGR rate measurement includes a substantial measurement delay due to the long gas sampling line used as part of measurement setup.

The choice to install the WRAF sensor at the pre-ETC location is to avoid the impact of flow pulsations in the intake plenum to the

WRAF sensor measurement. Pulsations induce fluctuations on the EGR measurement in a range of up to about 3% EGR rate, as shown in the Figure 2.

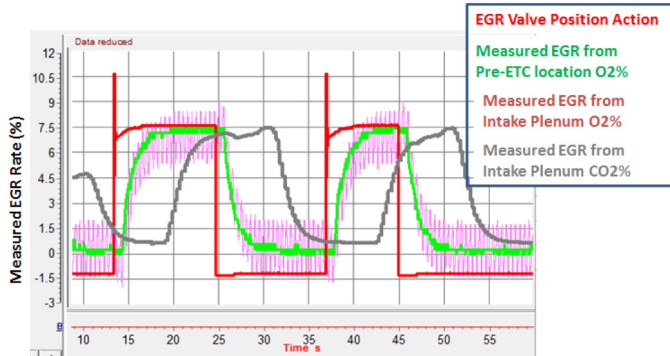


Figure 2. EGR measurement example (1500 RPM, 60 kPa MAP).

LOW-PRESSURE EGR ESTIMATION

The EGR estimation is part of the pneumatic and thermal state estimation logic [9] and is used for controlling the desired EGR fraction throughout the engine operation. It consists of three parts: 1) estimation of flow through the EGR valve, 2) estimation of the EGR valve outlet pressure, otherwise known as the mixing point pressure, and 3) estimation of the EGR valve outlet temperature, or mixing point temperature.

EGR Flow Estimation

The flow through the EGR valve is based on the compressible flow equation for flow through a valve [8]:

$$\dot{m} = \frac{P_{in} \cdot A_{eff}}{\sqrt{RT_{in}}} \cdot \varphi \left(\frac{P_{out}}{P_{in}} \right) \quad (3)$$

Where φ is a function of the valve pressure ratio (pressure ratio correction factor) [8, 10]. A_{eff} is the effective valve opening area, a function of the valve position and a discharge coefficient. Equation 3 is implemented by defining a flow at sonic standard conditions, and the flow equation is modified to the following form for the EGR valve flow estimation:

$$\dot{m}_{EGR} = \dot{m}_{std,EGR}(\theta_{EGR})|_{\varphi=const} \cdot \frac{P_{in,EGR}}{P_{std} \cdot \sqrt{\frac{T_{in,EGR}}{T_{std}}}} \cdot \varphi^* \left(\frac{P_{out}}{P_{in,EGR}} \right) \quad (4)$$

Where the function φ^* is a fixed relationship in software for the normalized pressure ratio correction, and $\dot{m}_{std}$ is a calibration table for valve flow at sonic standard conditions, a function of the valve position, θ_{EGR} . Figure 3 shows the flow characteristic of the EGR valve used in this test.

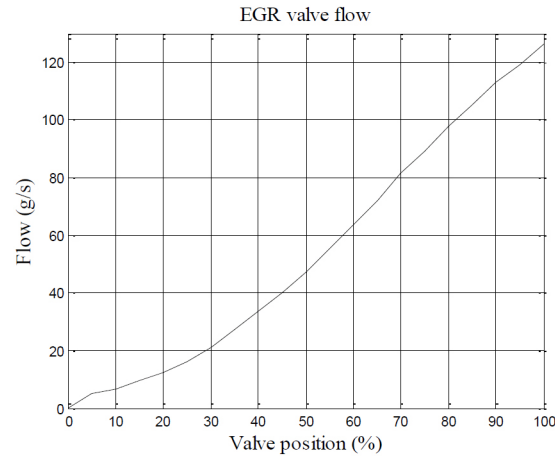


Figure 3. Example of the sonic standard EGR valve flow

The difference between EGR inlet and outlet pressures ($P_{out} - P_{in,EGR}$) is measured. The EGR valve inlet temperature ($T_{in,EGR}$) is measured. The flow through the EGR valve is used together with the measured fresh airflow to calculate the EGR fraction:

$$f_{EGR} = \frac{\dot{m}_{EGR}}{\dot{m}_{air} + \dot{m}_{EGR}} \quad (5)$$

Mixing Point Pressure Estimation

The key unknown parameter in Equation 3 is the EGR valve outlet pressure, P_{out} . Equation 3 is again used to calculate this pressure, but applied to the inlet mixing valve (IMV). In this case the equation is used to determine an unknown outlet pressure from a known inlet pressure, temperature and valve flow. The same principle is used in [10] to determine an unknown inlet pressure.

$$P_{out,EGR} = P_{out,IMV} = P_{in,IMV} \cdot K(\dot{m}, \theta, P_{in}, T_{in})_{IMV} \quad (6)$$

The input pressure, $P_{in,IMV}$, is the air filter outlet pressure, which is a calibratable pressure drop from barometric pressure. The parameter K is a calibration table that contains the pressure ratio across the mixing valve. The table is a function of valve opening area and a flow term that contains $\dot{m}$, P_{in} , and T_{in} . Figure 4 shows an example of this calibration.

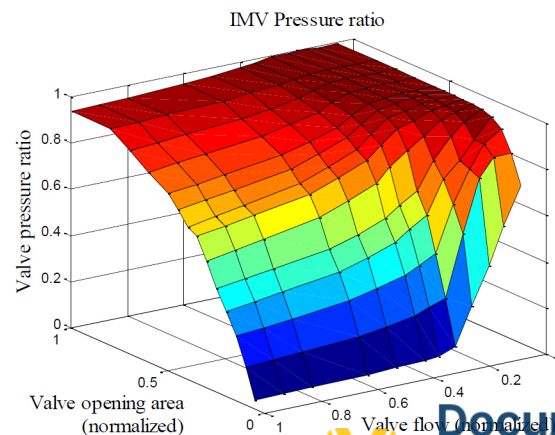


Figure 4. IMV Pressure Ratio.